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A review of waste heat recovery on two-stroke IC engine aboard ships

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ABSTRACT

Different types of waste heat recovery technologies available onboard ships have been discussed from the perspective of technical principle and application feasibility. Study of basic principle, novel methods, existing designs, theoretical and experimental analyses, economics and feasibility are discussed in this paper. The primary focus of this paper is to provide a better understanding of the options available for waste heat recovery and using in various applications onboard ocean-going ships to improve fuel economy and environmental compliance.

Contents

Abbreviations: WHR, waste heat recovery; HCCI, homogeneous charge compression ignition; MW, megawatt; SMCR, specified maximum continuous rating; kW, kilowatt; CO_2, carbon dioxide; TEG, thermoelectric generation; MED, multiple effect distillation; MSF, multi-stage flash; VTG, variable turbine geometry; VGT, variable geometry turbocharger; MIMO, multi-input multi-output; VTA, variable turbine area; SFOC, specific fuel oil consumption; LP, low pressure; HP, high pressure; COP, coefficient of performance; SCP, specific cooling power; PEPG, piezoelectric power generation; ORC, organic Rankine cycle; HRSG, heat recovery steam generator; MD, membrane distillation; HDH, humidification/dehumidification; TCS, turbo compound system; PTG, power turbine generator; TEU, twenty equivalent unit; CFD, computational fluid dynamics

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1. Introduction

The economic globalization has promoted the international flow of trade, investment, technology and financial capital, as a result of which, the globalization of the international shipping market has been accelerated further. Today the majority of prime movers (propulsion configuration) and auxiliary plants of ocean-going ships are diesel engines. In terms of the maximum power of installed engine of all ships in Germany above 100 gross tons, 96 is produced by diesel engines [1]. High-pressure combustion engine is still the basic propeller for ships due to both the most inexpensive heavy oil and the highest efficiency compared with all other heat engines. However, irreversibility in energy conversion is unavoidable by the second law of thermodynamics. While sailing in water, diesel engines onboard have an efficiency of about 48–51 and the rest of the input energy is discharged in the atmosphere in terms of exhaust gas and jacket water [2]. Much work now in progress is directed to the improvement of the thermal efficiency by optimizing the configuration of engine to achieve a better fuel consumption [3,4]. Also, much attention has focused on the advance combustion technologies, such as HCCI [5,6], lean combustion [7], stratified combustion [8,9], to achieve a higher overall efficiency and to reduce emission. However, as these technologies have achieved matured stage, it becomes harder and harder to get further improvement by using these methods. A valuable alternative approach to improving overall energy efficiency is to capture and reclaim the “waste heat”. Waste heat recovery system is one of the best energy saving methods to make a more efficient usage of fuels to achieve environmental improvement. Unlike the automobile operating conditions, the engine of ship especially that of large tonnage ship runs at a constant speed for a long time. It is easier to make use of more stable waste heat on ships compared with that of automobile. Furthermore, it can provide both heat source (waste heat) and cooling source (sea water).

From the aspect of environment, emissions of exhaust gases and particles from seagoing ships contribute significantly to the anthropogenic burden. In order to protect the Earth’s climate and environment and alleviate the energy crisis, extra effort is made to design Green Ship in future. Although clean combustion technology and after-treatment technology are getting matured [10], it is still hard to meet the stringent emission rule. WHR will be an effective way to produce more power on the basis of the same emission quality. It is another reason why WHR technology attracts much more attention of both energy and environment researchers.

The merchant fleet all over the world represents almost 80 of all the vessels ordered each year. Among them, 85 are powered by two-stroke diesel engines with the remainder having four-stroke engines [11]. The two-stroke diesel engine possesses economical and operational benefits compared to others. And its low rotation speed makes a low friction and higher efficiency feasible. moreover, it burns the cheapest residual fuel.

The energy reclaimed from the engine depends to a great extent on the size of the main engine and trade pattern (main engine load and ambient temperatures) of the ship. The engine size, operation route, loading condition and environment should be taken into consideration before choosing an appropriate way to waste heat utilization. Before research on recovering waste heat from diesel engine, the analysis of energy balance should be carried out to find out the potential of WHR. Scappin [12] evaluated the performance of marine two-stroke diesel engines by means of an energy balance. He [13] carried out an analysis of energy balance and used combined cycles to recover energy from different waste heat sources in engine.

A study [14] estimating a typical two-stroke diesel engine of MAN B&W Diesel has discovered that 25.5 of the released energy is wasted through the exhaust at ISO ambient reference conditions at 100 SMCR, and 16.5 and 5.6 in terms of the air cooler and jacket water respectively. Assuming the average operation in service at 85 SMCR=58,344 kW in 280 days a year, 24 h per day, 31,726 t of heavy fuel will be lost through the exhaust gas, air cooler and jacket water. If partial energy contained in the waste heat can be converted to useful power, it would not only bring measurable advantages for improving fuel consumption but also for reducing CO_2 and other harmful exhaust emissions correspondingly.

Each waste heat stream is investigated in terms of its waste heat quantity (the approximate energy contained in the waste heat stream), quality (typical exhaust temperature; usually higher the temperature, higher the quality and more cost effective the heat recovery), current recovery technologies and practices, and barriers to heat recovery. In any heat recovery situation, it is essential to know the amount of the recoverable heat and also how it can be used. Energy content of waste heat streams is a function of composition, mass flow rate and temperature [15,16], and is evaluated based on the process energy consumption, typical temperatures, and mass balances. There are three main heat sources with significant potential to be recovered as shown in Fig. 1, exhaust gas, air cooler and jacket cooler. The maximum exhaust temperature produced by two-stroke engine onboard ship is relatively low compared with four-stroke diesel engine, in the range of 250–500 °C. However, the quantity is in large amount [15]. Therefore significant amount of energy stored in exhaust gas is attractive to be recovered. Another promising character is that the limitation of the WHR systems’ mass and volume are not as strict as that for vehicles. The scavenge air that enters the charger air cooler is between 130 and 150 °C, and between 70 and 120 °C for jacket cooling water at the engine outlet.

Therefore, to capture and reuse the waste heat onboard is an emission-free substitute for the costly purchased fuel. WHR can be used not only for environmental control purposes, but also for improving the efficiency of fuel consumption. The drastic reduction on power consumption will directly minimize requirements of fuel and ship gross weight, which also will increase the cruising

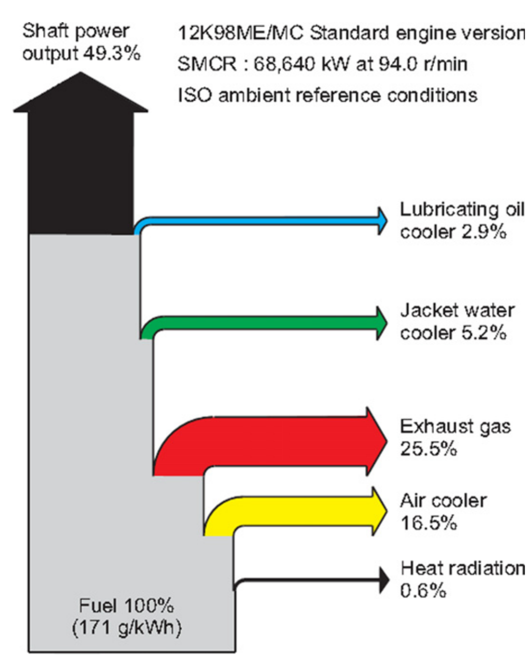

Fig. 1. Heat balance diagram of the nominally rated 12K98ME/MC engine of the standard engine version operating at ISO ambient reference conditions and at 100 SMCR.

range. An important number of solutions have been proposed to generate power, electricity and heating from the waste heat sources. As the flow rate of waste heat source aboard ships is in a large amount, the potential for waste heat recovery is particularly promising.

This paper is devoted to the WHR technologies that are available to convert low grade waste heat to useful forms or that have been already used aboard ships. The waste thermal energy on ships has been proposed to be used for space heating [17], heavy diesel fuel heating [18] and ballast water heating [19]. However, if the WHR system produces more other useful forms needed onboard than direct heating, fuel consumption and the sailing cost will achieve a further reduction. Whether a new technology is utilized in reality or not is mainly based on both performance and its economics. We take working principle, performance and economics into account for each recovery technology in this paper. Although Striling cycle engines have proven their capabilities to operate with waste heat, the complicated mechanical arrangement [20] and its transient response time [21] will be the practical barriers that hinder the development and adoption of Striling engines. Therefore, the heat-recovery possibilities taken into account in this paper are turbocharger/power turbine, fresh water obtained by using MED or MSF desalination technology, electricity/power obtained from Rankine cycle, air-conditioning and ice-making obtained by using sorption refrigeration, and combined WHR systems. All the technologies mentioned above are cost-effective ways to extract energy from the waste heat.

2. Main technical methods of WHR

At present, only a few of the ocean-going ships have already used WHR systems for direct thermal use, which use only a few portion of the waste heat energy. Energy crisis and the soaring fuel oil price have taken the concern of technologies to convert waste heat into useful energy.

An example of WHR would be that the high temperature stage was used for electric production or mechanical power, and the low temperature stage for process feed water heating or space heating. Due to the different characteristics and applied temperature ranges, different techniques must be selected according to both the heat source and the daily life requirements aboard ships. Technologies available to recover waste heat and to be served for daily needs aboard ships are discussed and studied as the following. Turbine, refrigeration, thermoelectric generation, desalination and Rankine cycle will be introduced in order.

2.1. Turbine

Turbine is a component that transfers enthalpy into kinetic energy. If the kinetic energy can be used to power a compressor, it can be called as a turbocharger. If it is used to power a generator or be combined into a power device, it may be called as a power turbine. Both the turbocharger and power turbine used simultaneously may be called as turbo-compounding.

2.1.1. Turbocharger

2.1.1.1. Principle and theory. In order to meet the regulation of the engine emission which is increasingly stringent, turbo charging has become the primary enabler for reducing emission and boosting fuel economy. Nowadays, almost all medium and large diesel engines are equipped with a turbocharger since it increases the mass of air entering the engine to improve both drivability and emissions from engines at the same time [22,23]. However, the applications of turbocharger also lead to higher cylinder back pressure, which may cause more exhaust gas remained in the cylinder during exhaust stroke. An optimized thermal efficiency can be achieved only when an appropriate turbo compression ratio is selected [34].

A turbocharger consists of a turbine and a compressor on a shared shaft. It converts the heat energy from the exhaust to power, which then drives the compressor to compress ambient air [24]. Normally, the air heated by the compression passes through a cooler which reduces its temperature and increases its density, and then is delivered to the air intake manifold of the engine at higher pressure. Thus, the amount of air entering the engine cylinders is greater, allowing more fuel to be burnt. As a consequence, the engine produces more power without increasing the engine size. Fig. 2 shows typical arrangement for a 4-sroke engine with turbo charging. Normally, the turbocharger used on 4-stroke engine is mainly radial type. However, axial type turbocharger is used on the 2-stroke engines for its high power output. The schematic diagram of a 2-stroke with and without scavenging air pump are both shown in Fig. 3.

2.1.1.2. Studies and performance. Turbo charging has played a vital role in the development of the diesel engine. The idea of supplying air at higher pressure to a diesel engine was proposed by Dr. Rudolf Diesel as early as 1896. And the use of a turbocharger of this purpose was the result of work by a Swiss, Alfred Buchi, whose idea was to use the exhaust gases of a diesel engine to drive a compressor via a turbine.

In recent years, the existing researches are mainly focused on the performance of turbocharger and benefits on diesel engines. Baines et al. [25] analyzed the heat energy transfer in automotive turbochargers and the results showed that the external heat transfer from the turbine accounts for approximately 70 of the total turbine heat transfer, and that the fraction of internal transfer to the lubrication oil is roughly 25, the remainder (about 5) to the compressor respectively. The recovered energy from turbine that drives the compressor is little and most is lost in atmosphere. Karabektas [26] compared the performance and exhaust emission characteristics in diesel engine for cases of naturally aspirated and turbocharged conditions and came to a conclusion that application of turbocharger can improve both thermal efficiency and CO emission. Theotokatos [27] presented a mathematical model of MAN B&W 6L60 engine and calculated the conservation of the turbo charging system, and derived an important result about the variation of the parameters of turbocharger. Weerasinghe [23] developed a mathematical model of turbo compounding. The simulation result showed that recover

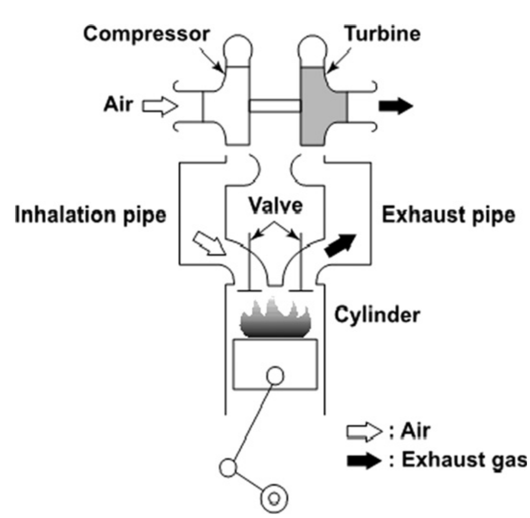

Fig. 2. Arrangement for 4-stroke engine with turbo charging.

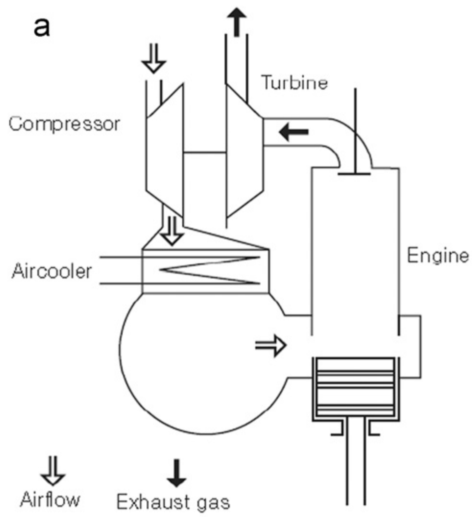

Fig. 3. Arrangements for turbocharged 2-stroke engine without and with scavenging air pump.

power contributes a maximum of 7.8, and the weight of the turbo compounding system is around 100 kg.

Variable geometry turbocharger (VGT) is one of the new turbine technologies that are getting matured. VGT is developed to precisely match the volume of charge air to the quantity of injected fuel at all points in the load and speed range of engines. VGT gives an extra degree-of-freedom in the propulsion control system which allows some amount of independence between engine speed and air-to-fuel ratio. This provides significant performance advantages: in steady-state operation the air-to-fuel ratio can be tuned independently of engine speed to improve efficiency. The authors of reference [29] assessed the feasibility and potential benefits of VGT diesel engine for transient ship maneuvers and emission control. With a multi-input multi-output (MIMO) controller, both torque and emission generation in marine diesel propulsion can be significantly improved in the simulation results. VTA (variable turbine area) is one form of VGT. According to the result gained from MAN Diesel & turbo [30], the reduction in SFOC on the engine fitted with VTA was as much as 4.4 g/kWh compared with the standard engine—or well over 2.5.

As the exhaust gas of multi-cylinder engine is not continuous during the operation time, exhaust pulse from different cylinders possesses a portion of energy. Multi-entry turbine is designed in order to isolate overlapping exhaust pulses from different engine cylinders. This design helps to reduce damping of the pressure peaks and ensures that the maximum possible amount of useful energy is delivered to the turbine wheel. Flow characteristic of double-entry turbine CFD module was calculated in Copeland's investigation [31]. The mass flow characteristic and efficiency characteristic of both equal and unequal admission module are predicted. Romagnoli [32] made comparison of the performance parameters between the three turbine configurations (nozzleless single-entry, variable geometry single and twin-entry). The results show the twin-entry configuration has better performance at high velocity ratio regions of the maps than single-entry configuration.

The two-stage turbocharging is adopted for the purpose of higher intake air pressure. As shown in Fig. 4, the two-stage turbocharger model is equipped with two parallel/series turbines and two series compressors, in which the compressed air from LP compressor outlet gets its higher compression ratio for the second compression in the HP compressor. A two-stage model based on GT-power was built by Xianfei and Bin [33]. The results showed that the two-stage turbocharger can satisfy the needed boost pressure of aircraft engine and ensure the power of engine be recovered to ground condition at altitude of 5–10 km that one-stage turbo charger cannot. Results obtained by Galindo et al. [34] proved that two-stage systems provide a difference up to 10 in terms of brake thermal efficiency at 2 bar of boost pressure. However, the difference exceeds 100 at 4 bar of boost pressure due to the difficulties for single stage system to achieve high compression ratio with good efficiency.

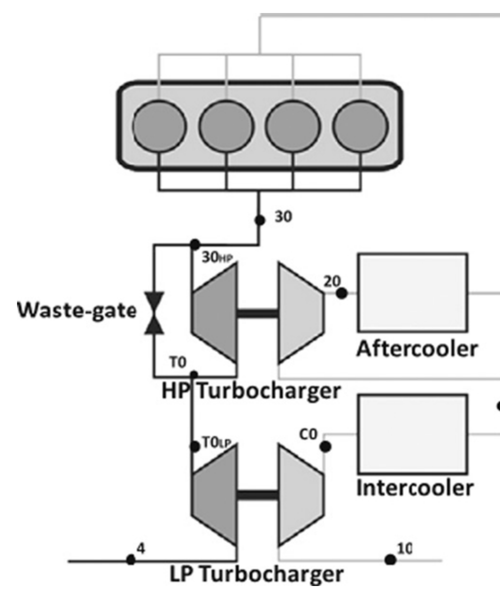

Fig. 4. Two-stage turbocharging scheme.

Fig. 5 shows an electric turbo compound system diagram proposed by Caterpillar [35]. We can see that one turbine in this system provides power for the requirement of both the compressor and a generator with the same shaft. It is different from those systems that consisted of a turbocharger plus an additional power turbine. Therefore, it is more simple and compact from the aspect of the structure. The fuel consumption was predicted to be reduced by 5–10 with electric turbocompound. However, more complicated control strategy has become its main disadvantage.

2.1.1.3. Economics and feasibility. Application of turbocharger on ships is popular around the world. Different from mechanical

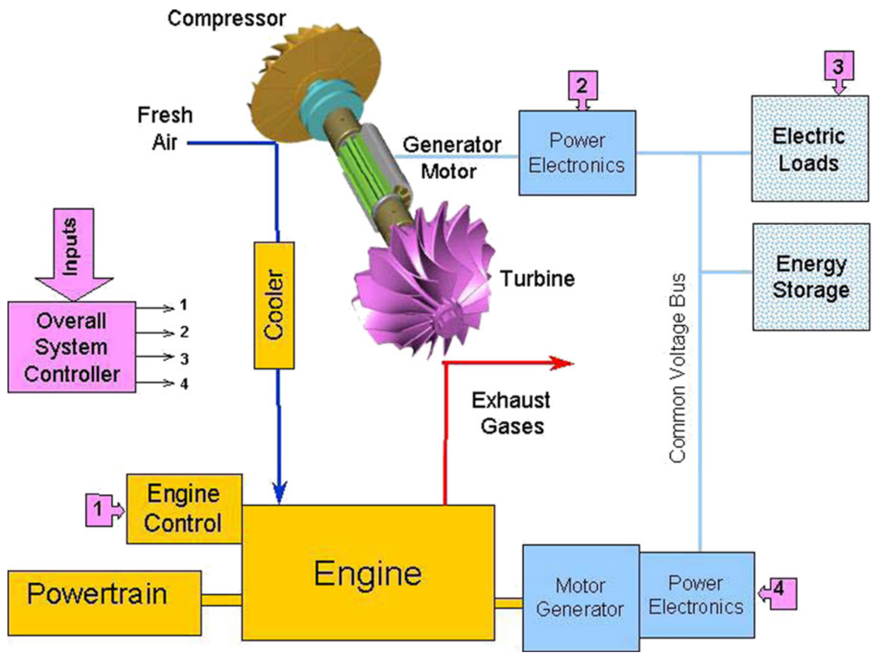

Fig. 5. Schematic of electric turbo compound.

charging, the original purpose of turbocharger is to increase the mass flow of air into the cylinder to improve the combustion by using the energy in the exhaust gas. From the aspect of performance, taking the diesel engine as an open thermal cycle, the turbo charging method is available to reduce the energy in the outlet and increase the overall efficiency. For example, a case study was presented in MAN Diesel & Turbo's own publication [30] and the fuel saving lead by VTA reached 150 t per year for tanker Stena President with 6S46MC-C engine of 7860 kW. Assuming that the price of heavy fuel oil is $700/t, the returns will be over $100,000 per year. It reduces the operation and maintenance cost significantly for the ship owners and brings them an appreciable interest. In addition, the investment of marine auxiliaries can be ignored.

2.1.2. Power turbine

2.1.2.1. Principle and theory. Due to high efficiency of the turbocharger, only a small portion of the exhaust gas is sufficient to secure the power needed for compressing the charging air. For the purpose of further utilization energy in exhaust gas, the application of power turbine is appreciated in a WHR system. Power turbine can implement continuous exhaust heat utilization for heavy duty engines, especially aboard large tonnage ships. In a WHR system, power turbine is used as a thermal propulsion device for improving the fuel efficiency of the main engine on ships. Compared with steam power plants, power turbine is characterized by its relatively low capital cost. It also has environmental advantages and short construction lead time.

There are two arrangements based on the location of power turbine. In the first variant the diesel engine feeds in parallel the turbocharger and the power turbine with the exhaust gas taken from the main exhaust gas manifold. In the second variant, the turbocharger and the power turbine are fed in series from the diesel engine exhaust gas manifold. The power turbine located in the downstream of the turbocharger. Both arrangements use a bypass valve to direct the distribution of the exhaust gas flux.

It has to be mentioned that the output of gas power turbine depends to a large extent on the exhaust mass flow rate the turbocharger needed, which is a main influence factor of the turbocharger's efficiency. For partial loads, the exhaust gas flow from the main engine may not be sufficient to secure the additional operation of the power turbine [2]. Before the application of power turbine, we must calculate and find out how much exhaust gas is needed to satisfy the turbocharger, to ensure the gain in reclaimed energy more than a compensation for the loss in efficiency from the higher fuel consumption. Therefore, the control of the exhaust distribution between turbocharger and power turbine is a main problem we have to solve. Although the power turbine has no adverse influence on the turbocharger, it increases the complexity of the control strategy.

2.1.2.2. Studies and performance. The energy reclaimed from exhaust gas by power turbine can be fed directly to the engine shaft, allowing the engine to be run at a correspondingly reduced output and to deliver the same power to the propeller shaft, or power a generator to provide electric for electrical equipment for meeting general ship demand. Both applications can save the fuel consumption and increase the voyage distance.

According to data from ABB Company, ABB Turbo Systems deliver more than 130 power turbines with electrical power of up to 1200 KW. ABB Company also proposes the application of variable turbine geometry (VTG) as power turbine to allow optimal matching of the main engine under variable operating conditions. New products were launched on the market under such names as "Efficiency Booster System (EBS)" by Sulzer of Winterthur or "Turbo Compound Systems (TCS)" by MAN B&W of Copenhagen. Dzida and Mucharski [2] came to a conclusion that for the 9RTA-96C Sulzer main engine produced by Wärtsilä, the operation of the power turbine is inadvisable when the main engine power is below 60–70.

2.1.2.3. Economics and feasibility. There is no paper about information on power turbine especially on its economics. So in order to evaluate economics and feasibility, we take 70,000 dwt Tanker

Stena President with 6S46MC-C engine of 7860 kW for example. According to literature [37], the recovery ratio of PTG only is 3–5. Assuming that the ship was operated at SMCR for 300 days per year, 1,697,760 kW of electricity would be generated by PTG. According to the internet, the price of electricity in America is about \$0.1674 per kWh, the annual saving of US\$284,205 would be attainable. As PTG does not need additional energy except for the exhaust gas, the M&O cost is low and the main contribution of the cost is the installation investment.

2.2. Refrigeration

Refrigeration devices are of significance to meet the needs for cooling requirements such as air-conditioning, ice-making and medical or food preservation. As the conventional mechanical refrigeration consumes the precious fuel or needs electricity to achieve either air conditioning or ice making, the utilization of waste heat energy as heat source is promising for the refrigeration process. At present, some effort has been devoted to the utilization of the vast amount of the waste energy of diesel engine aboard ships for refrigeration.

There are several types of refrigeration technology being used on marine ship, including compression refrigeration, sorption refrigeration and injection refrigeration. These technologies are not only used for air conditioning in summer, but are also used as icemaker for fishing vessels. However, it needs extra energy to drive the compression refrigeration system and injection refrigeration system, which leads to the increase of fuel consumption of ships. Sorption refrigeration system is driven by thermal energy and needs little electricity, which can utilize the waste heat of the engine and improve the energy converted efficiency. Therefore, considerable fuel can be saved and the mileage of the ship will increase. Another attractive feature is that a sorption refrigeration system is almost noise-free and virtually maintenance-free [38]. Both absorption refrigeration and adsorption refrigeration belong to the sorption refrigeration.

2.2.1. Absorption refrigeration

2.2.1.1. Principle and theory. The technology to produce a chilled effect from heat sources, which is called absorption chiller, is based on two fluids (refrigerant and absorbent), mixed and separated continuously. The heat source transfers energy to the strong solution and separates refrigerant and absorbent, while the refrigerant uptakes the heat from external environment during evaporation and makes the temperature of environment lower.

2.2.1.2. Studies and performance. Absorption refrigeration was discovered by Nairn in 1777, though the first commercial refrigerator was only built and patented in 1823 by Ferdinand Carré [39]. Many recent efforts have focused on the development of absorption cycles for their applicability to low grade heat source. Kececiler et al. [40] proposed that the heat source of temperature from 50 °C to 200 °C is economically attractive for absorption Refrigeration Systems. Fernandez-Seara et al. [41] carried out an investigation on an ammonia–water absorption refrigeration plant for onboard cooling production, which required a heat power range from 100 to 150 °C. Examples of current installed systems include those that use source temperature of 149 °C [42] and 143 °C [43], or even lower temperature of 68–93 °C from geothermal and solar application [44]. These studies indicate that waste heat sources aboard ship, such as exhaust gas, scavenge air and jacket cooling water, are all potential to drive an absorption refrigeration plant.

Among all the working fluids, ammonia refrigerant–water absorbent (NH_3–H_2O) and water refrigerant–lithium bromide absorbent (H_2O–LiBr) are the most popular ones in application. Lithium bromide–water systems are fairly well developed and have already been in use for many years but water–ammonia systems have a considerable scope for improvement and applications in many countries. Research results have shown the feasibility of employing a gas-to-thermal fluid heat recovery system to power a NH_3–H_2O absorption refrigeration system for the use in trawler chiller fishing vessels [38].

Manzela [39] described an absorption refrigeration system driven by means of engine exhaust gas. Srikhirin et al. [45] reviewed a number of research options of absorption refrigeration technology and compared various types of absorption refrigeration systems. Another recent research is the work of Fernandez-Seara [41], who tried to control the exhaust gas flow rate by means of a bypass valve. For this system, the liquid-to-solution heat exchanger acts as a generator and the gas-to-liquid heat exchangers as an economizer. This prototype can save from 2 to 4 of the total ship fuel consumption. An experiment research by Kececiler et al. [40] showed that when the mass flow rate of hot water from the Hot Spring at 60 °C is 12.5 kg/s, it is sufficient for the absorption refrigeration System operating on water–lithium bromide to produce 225.57 kW cooling effect. The result of this research provides the information that the jacket water of marine engine is sufficient to satisfy the requirement of this water–lithium bromide cooling system.

In order to enhance the value of coefficient of performance (COP), various new absorption cycles have been developed based on the basic absorption system. For the purpose of extracting 1 kW of energy from a waste heat stream at a nominal 120 °C, Little and Garimella [46] investigated and compared 5 different systems that generate cooling, higher-grade heating, or mechanical work. The simulation results showed that the absorption cooling cycle works with much higher COPs ($>\sim 0.7$). Combined ejector–absorption refrigeration cycles [47,48] were proposed to improve the performance of the absorption refrigeration cycle. These cycles have a higher COP than that of the conventional single-effect cycle when the temperature of heat source is higher than 130 °C.

For the purpose of further reduction of electricity or power, a novel system (Fig. 6) with an expander–compressor proposed by Hong et al. [49] showed that the COP is higher than that of the conventional single effect cycle even when the conventional two stage cycle operates at most conditions due to the higher absorption pressure of new cycle without pump work input. A schematic of the cascaded absorption/vapor-compression cycle is shown in Fig. 7 [50]. This system model was powered by exhaust heat, and characterized by its two cycles in order to meet different cooling purposes, Libr–H_2O absorption cooling cycle for medium-temperature coolant (~ 5 °C), CO_2 vapor compression system for low-temperature (-40 °C). Compared to an equivalent vapor-compression system, the cascaded absorption/vapor-compression cycle avoids up to 31 electricity demand.

2.2.1.3. Economics and feasibility. Misra et al. [51] and Kizilkan et al. [52] applied thermoeconomic theory to evaluate the economic cost of LiBr absorption refrigeration system, and investigated how to make a thermoeconomic optimization. An economic model of LiBr–H_2O absorption refrigeration system was developed by Rubio-Maya et al. [53] to optimize the annual operating cost. The values of annual cost of operation and investment were \$14,993 per year and \$41,718 respectively, which were decreased in about \$150 per year and \$19,012 respectively compared with the earlier reference [54]. A case of Misra et al. [54] proposed that unit cost of input exergy is 0.03785 \$/kWh for a system with cooling capacity of 201.29 kW.

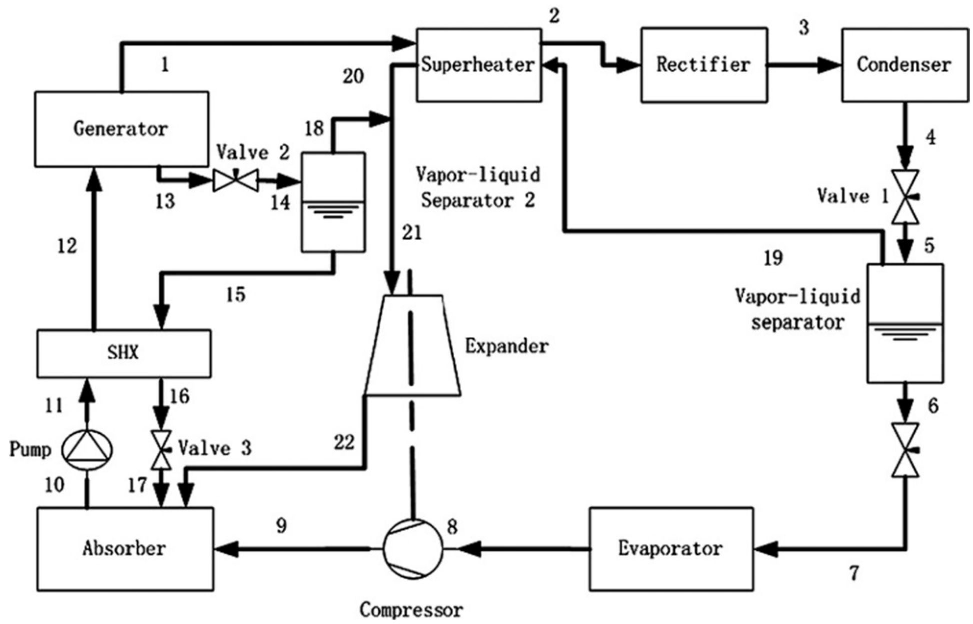

Fig. 6. Schematic diagram of the novel absorption refrigeration cycle with an expander–compressor [49].

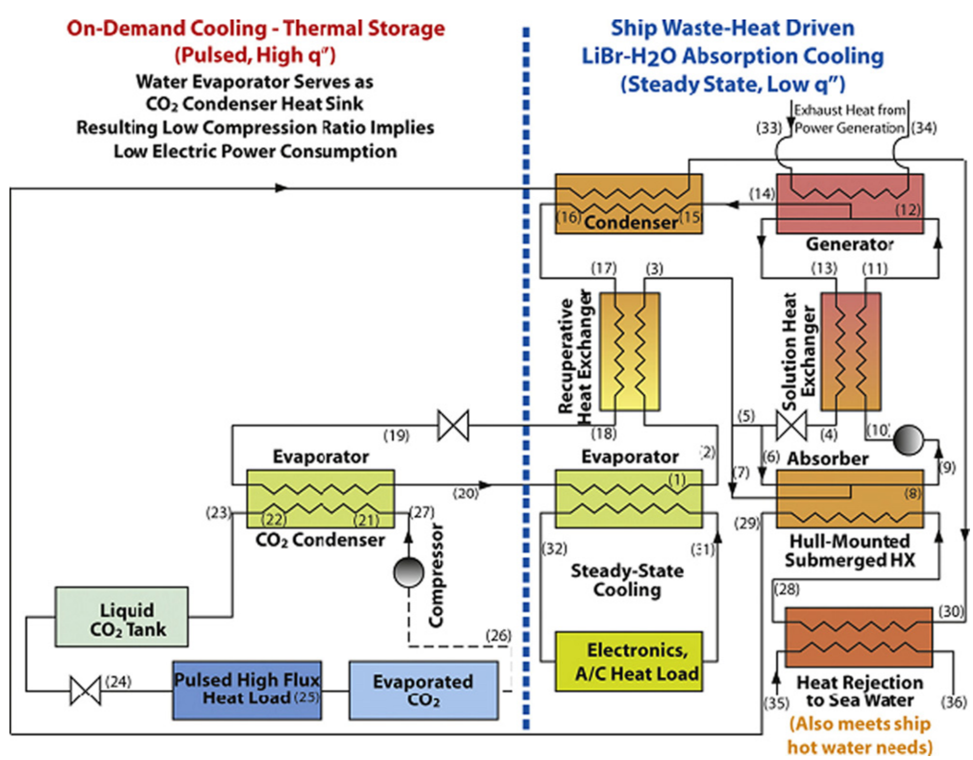

Fig. 7. Schematic of cascade absorption/vapor-compression cycle.

For the absorption systems driven by waste heat on ships, assuming the energy of power output and exhaust to be 45 and 25 respectively when they are operating at the rated condition, a two-stroke engine of 10 MW work output contains 5.56 MW energy in exhaust gas. Supposing that the energy available to be recovered is 50, and the COP of the system is also 50 (according to [46], the COP is about 0.5 if a heat source temperature is higher than 150 °C), the cooling capacity reaches 1389 kW and it is available to provide air conditioning for more than 9000 m^2 if the requirement of air conditioning is 150 W/ m^2. Assuming the price of the electricity and time of refrigeration process were $0.1674 per kWh and 18 h per day respectively, $104,633 can be saved in a month.

2.2.2. Adsorption refrigeration

2.2.2.1. Principle and theory. There are two main materials in an adsorption refrigeration system, adsorbent and adsorbate. Adsorbents having special affinity with polar substances like water are termed "hydrophilic". These include Silica gel, Zeolites and porous or active Alumina. Non-polar adsorbents, termed

"hydrophobic", have more affinity for oils and gases than for water [55]. As explained by Ruthven [56] and Suzuki [57], adsorption occurs at the surface interface of two phases, heating–desorption–condensation phase in which the adsorbate was desorpted from the absorbent then condensed liquid adsorbate was transferred into the evaporator, and cooling–adsorption–evaporation phase in which the liquid adsorbate evaporates and makes cooling effect. Compared with an absorption system, the adsorption cooling system has the advantages of mechanical simplicity and high reliability [58].

2.2.2.2. Studies and performance. As shown in Tables 1–3, several refrigeration applications [59–63] have been studied using various adsorbent and adsorbate pairs. Zeolite and activated Carbon get rapid development and have become the most popular adsorbent in most systems. The typical adsorption refrigeration cycles include basic cycle, continuous heat recovery cycle [64], mass recovery cycle, thermal wave cycle [65], convective thermal wave cycle [66], cascade multi-effect cycle [67], hybrid heating and cooling cycle.

Recently, many investigations about simulations of adsorption cycles [68–70] have been reported, and different models have been developed to evaluate the thermal performance of the adsorption cooling systems in terms of the coefficient of performance (COP) and specific cooling power (SCP). Simulation study also provides a theoretical basis for the development of adsorption cooling systems. Y. Liu and K.C. Leong [69] present a new transient two-dimensional model to study the effect of mass flow rate of jacket water on system performance.

Some experiments of adsorption refrigeration have also been done. In Wang et al.'s literature [71], an adsorption system with activated Carbon and Methanol as working pairs was developed for ice production. As the heat source needed for the activated Carbon–Methanol system is about 80–110 °C, the jacket water can satisfy this temperature requirement. When the temperature is 100 °C, the refrigerator achieves a refrigeration power density of more than 2.6 kg ice per day per kg activated carbon with a COP of 0.13, for air conditioning with a COP of about 0.4. Tamainot-Telto and Critoph [72] presented the description of a laboratory prototype of an adsorption cooling machine, which used an activated monolithic Carbon–Ammonia pair and the steam boiler producing steam from 100 °C to 150 °C as the heat source.

Several researches [60,63,73] have been done in the study of adsorption air conditioning powered by the waste heat energy, of which most system used Zeolite–water as working pair. Generally, the temperature of diesel engine exhaust gas is higher than 250 °C, which can satisfy the requirement of heat source for an adsorption refrigeration plant. The condenser can be chilled by the seawater, the temperature of which is about 5 °C. In Lu et al.'s research [73], an adsorption driven by exhaust gas from a diesel locomotive system, which incorporates one adsorbent bed and utilizes Zeolite–water as the working pair, is feasible to be applied for space conditioning of the locomotive driver's cab. Another experiment done by Lu et al. [64] with Zeolite–water as the working pair showed that the COP of the system was 0.38, which could meet the demand for a practical automobile waste heat adsorption cooling system (mobile), and the SCP was 25.7 W/kg.

The main obstacle of the development of adsorption refrigeration technology is its low coefficient of performance. Researches on how to improve the performance of adsorption cooling systems have been investigated and a way to enhance the parameters in terms of COP and SCP was found. The refrigeration performance can be improved by improving the mass transfer

Table 3
Performance of different systems [37].

Waste heat recovery system recovery ratios	
Configuration	Efficiency as of main-engine SMCR (depending on size)
PTG	3–5
Single steam pressure-STG	4–7
Dual steam pressure-STG	6–9
Dual steam pressure steam and power turbine unit	9–12

Table 1
Several studies about adsorption refrigeration.

References	Working pair	Type	Desorpted temperature	COP	SCP (W/kg)
[59]	Zeolite–water	Air conditioning	150 °C	0.25	7
[60]	Zeolite–water	Air conditioning		0.38	25.7
[61]	Activated carbon–methanol	Air conditioning	100	0.4	150
		Ice making		0.13	2.6 kg/kg
[62]	Consolidated activated carbon–methanol	Refrigeration		0.125	32.6
	Consolidated composite adsorbent-ammonia	Refrigeration		0.35	493.2
[63]	Zeolite–water	Chilled water	450	0.25	200
[64]	$AC+CaCl_2+NH_3$	Ice making	115	0.39	770

Table 2
Comparisons of MSF and MED.

Desalination type	MSF	MED
Feature	Bulk liquid boiling	The steam generated in one stage is used to heat the salt solution in the next stage
Temperature	Around 100 °C[102]	55–90 °C [110,117]
Production [108]	$23–528\times10^3$ m^3/day	$91–320\times10^3$ m^3/day
Energy consumption [112]	18 kWh/m^3	15 kWh/m^3
Capital cost [108]	0.52–1.01 $/$m^3$	0.52–1.75/m^3
Capacity	Al-Jubail in Saudi Arabia, the biggest MSF plant in the world, with a capacity of 815,120 m^3/day; Shuweiat plant, with a capacity of 75,700 m^3/day [104]	He Umm al Nar MED plant with a unit capacity of 15, 911 m3/day; Sharjah plant, with a unit capacity of 22,730 m^3/day; another example, in 2000 in Las Palmas Spain, with a capacity of 17,500 m3/day [111]; Reliance Refinery (India)—4 × MED 12,000 m^3/day [110]

performance and reasonable design [74]. A research of Li et al. [75] proposed a combined double-way cycle, which could improve COP by 167 and 60 when compared with conventional adsorption cycle and resorption cycle respectively.

2.2.2.3. Economics and feasibility. Adsorption technologies for refrigeration using waste heat source is only researched in the current decade. Although the solid–vapor adsorption systems are still in the experiment state, researches have shown that it has a promising potential for competing with conventional vapor-compression technologies. An estimation of the cost of an adsorption air conditioning system using Zeolite and water is about 5000US$ [73]. The average refrigeration power for this system is 3.75 kW, the installation cost of which is about 1335US$ per kW cooling power. As the heat energy from internal engine is free, the total cost of cooling will be the cost of both adsorption chiller and the operation cost of the pump. Certainly, the installation cost of such a system, which is still in testing phase, will achieve a significant reduction in commercial production.

Both absorption and adsorption cooling systems have their own characteristics and advantage, and both can be powered by waste heat energy. However, the work to increase the COP becomes the priority for the further development and application in future.

2.3. Thermoelectric generation

2.3.1. Principle and theory

Thermoelectric modules are solid-state devices that directly convert thermal energy into electrical energy. This process is based on the "Seebeck effect", which is the appearance of an electrical voltage caused by a temperature gradient across a material. The simplest TEG consists of a thermocouple consisting of n-type (materials with excess electrons) and p-type (materials with deficit electrons) elements connected electrically in series and thermally in parallel. Heat is input on one side and rejected from the other side, generating a voltage across the TE couple. A simple TEG package block diagram is presented in Fig. 8.

The efficiency can be expressed as a function of the temperature over which it is operated and so called 'goodness factor' or thermoelectric figure-of-merit of the thermocouple material Z

$$Z = \frac{\alpha^2 \sigma}{\lambda}$$

where $\alpha^2\sigma$ is referred to as the electrical power factor, with α the Seebeck coefficient, σ the electrical conductivity and λ the total thermal conductivity.

Thermoelectric phenomena result from the diffusion of electrons and phonons along a temperature gradient in electrically conducting solids. These diffusion currents are determined by the concentration of these particles, and their interaction with each other as well as with impurities and defects.

2.3.2. Studies and performance

Several studies on TEG in locomotive engines [76] and industry [77,78] applications have shown promising results. This property makes it possible to produce direct current electricity on board a ship by applying waste heat of exhaust gas on one side of a TE material, while exposing the other side to lower temperature sea water. It has no moving components, and is silent, totally scalable and extremely reliable.

The use of TEGs in heat recovery applications has been a major theme in the development of the field since the 1990s [76,77]. Kajikawa [79] reviewed the development TEG and gave an overview on the R&D of the thermoelectric systems. In mid-1993, tests of the 1 kW generators started in the Diesel engine test cell at Golden West College [80]. An example of thermoelectric generator powered by waste heat was mentioned in David's paper [81]. The generator converts about 5 of the input heat to electrical power, the remainder of 95 transfers to the hot water exchanger. It can generate 50 W when operated at hot and cold side temperatures of 550 °C and 50 °C. The authors of literature [82] mentioned that the resources with temperature lower than 300 °F is not suitable for TEG energy recovery. However, new TE materials potentially provide conversion efficiencies of 15–25 when operating at hot-side temperatures of 450–750 °C.

Conclusively, the conversion efficiency of TEG is low. However, TEG can be applied when coupled with ORC and it cools down exhaust temperature to ensure that the organic working fluid is under its decomposition temperature [83].

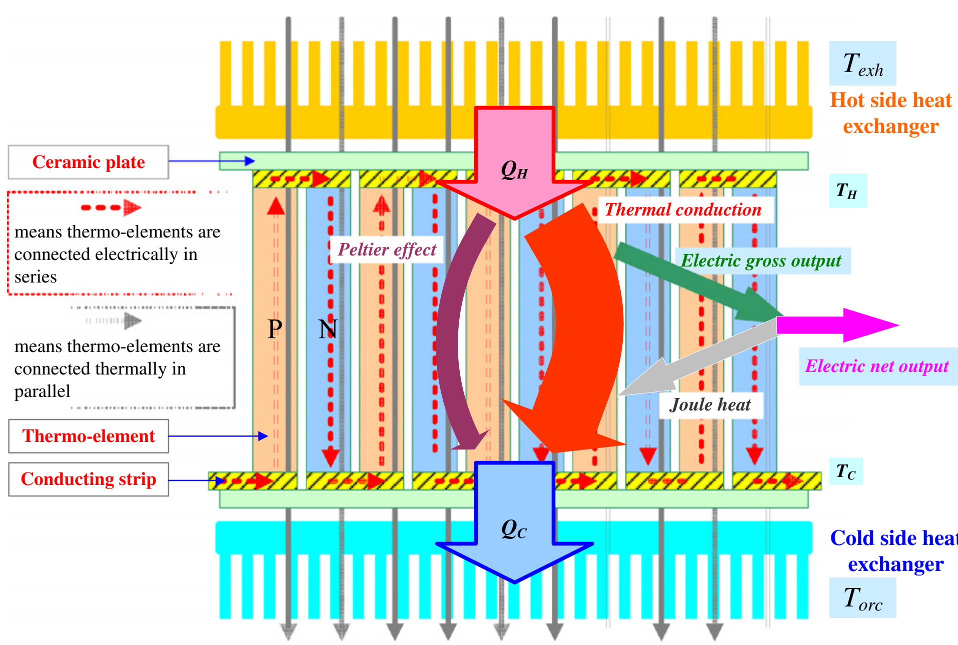

Fig. 8. Functional diagram of TEG.

2.3.3. Economics and feasibility

The performance of the thermoelectric materials has a significant effect on the cost of TEG recovery system. Early study on the cost of TEG carried out by Haidar and Ghojel [84] proposed that a full TEG WHR system was still prohibitive in 2001 due to its current price of US$175 per module for small orders despite the fact that the energy source is free and 2–5 of fuel consumption by vehicle can be saved with corresponding decrease in all pollutants including greenhouse gases. However, the development of materials has led to a significant reduction in the TEG cost. A cost analysis based on WATT-100 was carried out by David [85] which pointed out that the payback time is around 2 years for improved modules developed at Cardiff, where the consumer's purchase price of electricity is around £0.068 per kWh.

As the conversion efficiency is relatively low, the system can only become cost effective and more competitive when lower cost of TG modules and higher value of figure-of-merit can be achieved. Furthermore, it is more feasible for the combination of TEG and ORC. Due to the special environment on ships which can provide seawater as the cold resource, utilization of TEG is more potential on vehicles.

2.4. Rankine cycle

2.4.1. Principle and theory

Rankine cycle is a cycle that converts heat into work. As shown in Fig. 9, there are four main components in a basic Rankine cycle, steam turbine, condenser, circulation pump and evaporator. The working fluid usually circulates in a closed cycle. Firstly, working fluid from the outlet of the condenser is pumped into an evaporator. The exhaust gas waste heat is used to heat up steam for a turbo generator, in which energy is transferred from the boiler to the turbine through high pressure steam. The power produced by steam turbine will be transferred to the electric generator or output shaft. At last, the steam from the turbine outlet is condensed into fluid again in the condenser and a new cycle begins.

2.4.2. Studies and performance

Researches of RC began in the 1970s and its rapid development was obtained in recent years because of energy crisis and environment concerns. The current investigation on RC is concerned with how to improve the recovery efficiency by comparing different configurations and working fluid. Wang et al. [86] presented a review about thermal exhaust heat recovery of Rankine cycle, including effect of different system configurations, working fluids and components on the efficiency. The exhaust gas of internal engine on automobile works as a heat source, which is higher than that of on the ship we investigate. The configuration of exhaust gas only and the configuration of exhaust gas plus coolant have both been analyzed by BMW Group [87]. The simulation results showed that water would be a preferable working fluid for the configuration of exhaust gas only. If a low temperature heat source is used in addition, alcohol (e.g. Ethanol) would be more promising. Therefore, the selection of working fluid is based on the heat resource at a great extent. Parametric analysis of efficiency should be carried out before selecting the system configuration and working fluid.

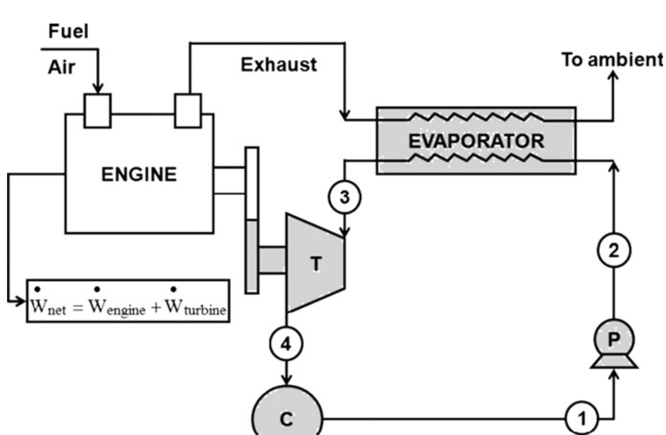

Fig. 9. Schematic of a basic Rankine cycle driven by exhaust of engine.

The organic Rankine cycle (ORC) uses an organic fluid such as n-pentane or toluene in place of water and steam. Organic fluid has gained more attention because it allows the use of lower-temperature heat sources. Hung et al. [88] analyzed efficiencies of ORCs using different fluids and came to a conclusion that isentropic fluids are the most suitable for recovering low-temperature waste heat. Bombarda et al. [89] make a comparison of Kalina and ORC cycles and found that the adoption of ORC cycles is superior to Kalina due to its simpler plant scheme, smaller surface heat exchangers and lower pressure to obtain the same electric power.

In summary, the selection of the working fluid and system configuration is critical to achieve high thermal efficiencies as well as optimum utilization of the available heat source. The temperature of exhaust gas aboard is about 200–500 °C, and 70–90 °C for jacket water. Both available resources for WHR are moderate and of low grade energy. In order to prevent the corrosion problem due to the low evaporation temperature of water–steam, it is worthwhile employing organic fluid in WHR for low-temperature heat sources, which would evaporate at a lower temperature than water–steam phase change. In an Organic Rankine cycle, less heat is required to vaporize the working fluid. Another main reason explained by Larjola [90] is that the best efficiency and the highest power output are usually obtained by using a suitable organic fluid instead of water in the Rankine cycle by recovering energy from moderate temperature heat sources. Although water or steam is still the main working fluid of RC systems on ships, application of organic fluid possesses a great potential in the future.

2.4.3. Feasibility and economics

The evaluation of cost of the RC system must take both the investment and maintenance cost into consideration. Moreover, the RC system should be evaluated by its net output power, the value that the power produced by the turbine subtracts the power consumed by the fluid pump. An evaluation method of the RC system was introduced by Gewald [91]. Economic analyses of steam turbine power plants with a power turbine topping cycle was carried out by Ringler et al. [87], the results revealed that significant fuel savings justify the capital investment. Leibowitz et al. [93] carried out a review of possibility to produce power output at the range of 20–50 kW by reusing low grade heat sources. The utilization of screw expander instead of turbines is better as the installation cost becomes lower than that of the conventional ORC systems, in the range of $1200–2000/kW of net power output.

Waste-heat-driven steam power plants are often not economical for capacities below 5 MW and for low-temperature waste-heat streams. For this reason, Brasz et al. [94] proposed that the utilization of ORC power plant hardware derived from air-conditioning equipment overcomes high cost problem since air-conditioning hardware has a cost structure almost an order of magnitude smaller than that of traditional power generating equipment. For example, the cost of air-conditioning is available at a cost of around $200–300 per kW.

However, the equipment cost of multi-megawatt conventional power generation equipment is about $1200–1500 per kW generator output or even higher for smaller power generation equipment.

Many researches [95–97] have turned out that organic fluids have large potential to be used as working fluid in WHR systems. If the leakage problem can be solved thoroughly, the development and application of organic working fluid will be popular in WHR system aboard ship.

2.5. Desalination

As a ship sailing in sea, large quantities of fresh water are required to meet everyday life demand. If the ship carried all the water they needed, it would take a lot of space and reduce the load of the cargo. So it is advisable to take appropriate action to utilize seawater for fresh water production. There are many methods of abstracting fresh water from salt water such as thermal distillation, reverse osmosis, freezing and electrolysis [98]. All the desalination processes above consume energy. For the special environment on ships, the most accessible access is to reclaim waste heat of the main engine for fresh water production. Therefore, among all types of technology, thermal distillation that applied the waste heat from the exhaust gas and jacket water has been proposed. Both multi-stage flash (MSF) and multiple effect distillation (MED) belong to heat-operated type units.

2.5.1. MSF

Although other distillation processes start to find their way in the market, the Multistage Flash Desalination technology is still considered as the workhorse of the desalination industry. MSF has a market share of over 60 of the worldwide desalination market, and in the Middle East Area this share jumps to almost 80 [99]. The reason for such a strong position is the great reliability and proof of such a mature technology.

2.5.1.1. Principle and theory. As shown in Fig. 10, the MSF principle is based on raising the temperature so that the seawater flashes when subjected to a sudden pressure drop in the first stage of the plant. The flashed water vapor is then cooled and condensed by colder seawater flowing in tubes of the condenser to produce distillate. The unflashed brine passes from one stage to the next and transfers heat during this process so that the seawater can be evaporated repeatedly without adding more heat. The MSF operates from a positive pressure in the first stage to a high vacuum in the last stage [100]. One of the advantages for MSF is simple layout, which can save a lot of space for other cargoes. Furthermore, its significant reliable performance guarantees its popularity for a long time. This process requires an external steam supply, normally at a temperature around 100 °C. The maximum temperature is limited by the salt concentration to avoid scaling and this maximum temperature limits the performance of the process. A key design feature of MSF systems is bulk liquid boiling. This alleviates problems with scale formation on heat transfer tubes.

2.5.1.2. Studies and performance. MSF plants have been built since the 1950s [101]. A great amount of research and development has been carried out in recent years. Khawaji et al. [98] reviewed current status, practices and advances based on MSF distillation and RO process, and outlined future prospects for different desalination technologies. An example of MSF desalination unit using waste heat onboard was introduced in Ghirardo et al.'s investigation [102]. The temperature of evaporator was operated at 90 °C for double-effect units, and 50 °C for single-effect units. The production of fresh water for a single-effect unit would be 7.1 m^3/day when the inlet temperature of heat source was 60 °C, in which about 212 kW of heat was available, whereas that for a double-effect unit would be about 10 m^3/day under the hot source temperature of 100 °C, in which about 184 kW was available. The weight of such a desalination plant was estimated to be between 156 and 320 kg and volume was between 0.45 and 0.96 m^3 for the single-effect unit producing 7.1 m^3/day. So the specific evaporator heat consumption was 717 kWh/m^3 for single-effect desalination units and 442 kWh/m^3 for double-effect desalination units. Another similar investigation [103] showed that the weight of a 9084 m^3/day plant was about 1000 t whereas that of a 36,336 m^3/day plant was 2500 t.

Many countries especially middle east countries that are short of fresh water attach great importance to the development of desalination technology. Therefore, desalination technology gains rapid development and some large MSF units have been built in these countries. For example, the Shuweiat plant located in the United Arab Emirates is the largest MSF unit, with a capacity of 75,700 m^3/day [104].

2.5.1.3. Economics and feasibility. It has to be mentioned that for economic consideration, a wide range of technical parameters must be evaluated including seawater characteristics, product water quality, source of energy and consumption, plant size, plant reliability, concentrate disposal, space requirements, operation and maintenance aspects, etc. [104]. The total production cost was also given by Al-Juwayhel et al. [105] in 1992, $1.86/$m^3$ for MSF process. Borsani and Rebagliati [106] presented comparisons of some past examples with most recent installations both in terms of technical requirements and installation costs and the analysis results showed that the water cost produced by MSF

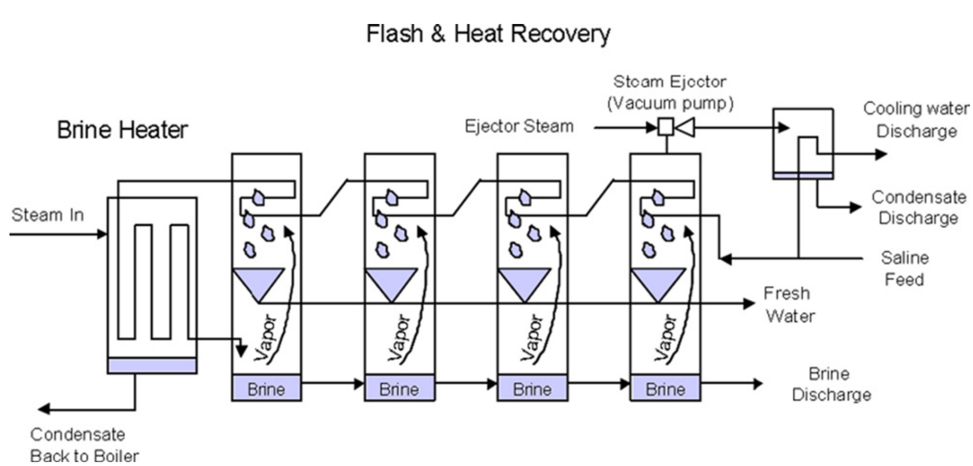

Fig. 10. Principle of operation of the multi-stage flash (MSF) system [116].

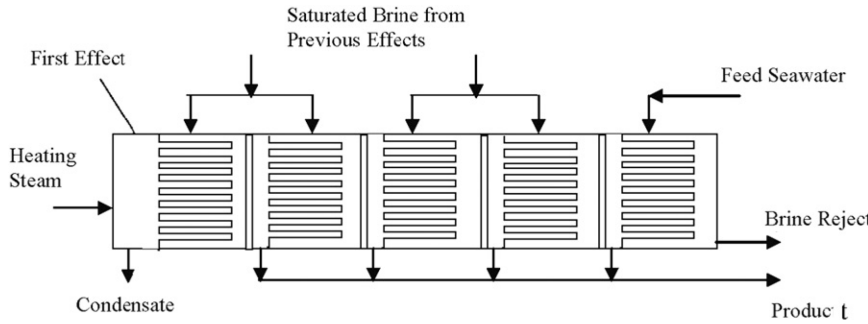

Fig. 11. Flow diagram of MED.

plants is almost in line with other technologies. Despite a cost increase in raw materials of more than 40 and a labor cost increase of more than 100 in the past 20 years, the price of selling MSF desalination plant showed a reduction of about 50. Another example [107] about a MSF process of capacity 528,000 m^3/day produces desalinated water at a cost of $0.42/$m^3$. The desalination cost can be reduced by 15 if the MSF unit is combined with a RO unit for the same production capacity. Another review carried out by Eltawil et al. [108] showed that for a MSF plant with a capacity of 7.13 million gallons/day, the capital cost is about 0.292 $Cent/gallon for the dual purpose, and 0.621 $Cent/gallon for the single purpose.

In summary, the reduction of capital cost makes the MSF plant of more potential for future use. The technical development will be the main factor which plays a significant role in its application and popularity in future market.

2.5.2. MED

2.5.2.1. Principle and theory. MED is also called multi-effect evaporation (MEE). As shown in Fig. 11, heating steam is fed to the first effect. This would result in the formation of a small amount of water vapor, which is used to heat the second effect. The vapor would release its latent heat and condense. The released latent heat would result in the formation of a smaller amount of vapor in the second effect. This process is repeated in subsequent effects, until the vapor temperature becomes close to the feed sea water temperature. It is necessary to increase the feed temperature to saturation temperature of each effect. MED limits the heat transfer process to release the latent heat from the heating vapor and the latent heat gained from the formed vapor. Therefore, a high energy can be reclaimed and the brine is rejected at a very low temperature.

2.5.2.2. Studies and performance. The first multiple effect desalination unit was installed in 1960. The main feature of the MED process is that it operates at a low top brine temperature between 60 and 70 °C [109], and even at a lower temperature of 55 °C. All of the waste heat sources aboard ships are capable to power a MED unit. By utilizing the waste heat energy from diesel engine, the only prime energy consumption is estimated to be 2.0 kWh/t used for the water pumps [110].

Ophir [110] studied some MED plants with the capacity 20,000–200,000 t/day and different features had been presented. Study on energy consumption was carried out by Ettouney [111] and Eltawil [108]. There was a significant difference in the energy consumption value due to the difference of the plant capacities and different base analysis parameters.

2.5.2.3. Economics and feasibility. The numerical value of the proposed desalination plant investment cost is a combination of the initial cost and the corresponding maintenance and operation (M&O) cost. Ophir and Lokiec [110] described the advantages of MED plants and analyzed how the advantages influence on the economics of the installation by reducing both capital and operation costs, increasing the availability and extending the life expectancy of the plant. Research on economics of MED was also carried out by Morin [112] and the results showed that the total production cost was $1.49/$m^3$ for the MED process in 1992. Since the development of technology and larger capacity of MED plant, the capital cost and operation cost have been reduced significantly.

Vlachos and Kaldellis [113] estimated the cost and fresh water production of a thermal desalination plant by using the exhaust gas of a gas turbine with a rated power of 43.3 MW. The annual production capacity of the desalination plant analyzed surpasses the 2,500,000 m^3 of potable water. The calculation results showed that the water production cost varies between 0.3 €/m^3 and 0.37 €/m^3 if the operation time is 5 years, whereas the corresponding value for the 15-years operation is between 0.18 €/m^3 and 0.25 €/m^3. The estimated price is much lower than the market mean price of 0.53 €/m^3. Another study [114] presented the analysis on the cost of MED plant. When using a dual-purpose type, the cost is $ 0.33 Cent/gallon for the capacity of 6 million gallons. However, the cost is $ 0.739 Cent/gallon for a single-purpose type of the same capacity, which overpasses twice compared with dual-purpose type.

In summary, comparison of cost between MSF and MED was presented in the following. According to Morin's literature [112], the MSF process needs about half the heat transfer surface area as much as that required for the MED process. The MED process offers a recovery of almost 50 higher than the MSF process for equal performance ratio. The total production cost was also given in Al-Juwayhel's paper [105], $1.86/$m^3$ for MSF process and $1.49/$m^3$ for the MED system. Good economics in construction, civil work and seawater intake result in costs 35 less than for MSF plants. Moreover, the MED desalination system is more efficient than the predominant MSF desalination system, from a thermodynamic and heat-transfer point of view. From these points, MED seems to be superior to MSF.

3. Combined cycle systems

The waste heat recovery technologies mentioned above are all available aboard ships. In order to make full use of the waste energy emitted from the engines, combination of two or more these technologies proves to be more attractive to ship owners. As the needed temperatures are different from each other, technology selection needs to be taken into consideration. In order to achieve an optimum strategy of recovering waste heat aboard ships, the optimum or just simple specific technology combination must be

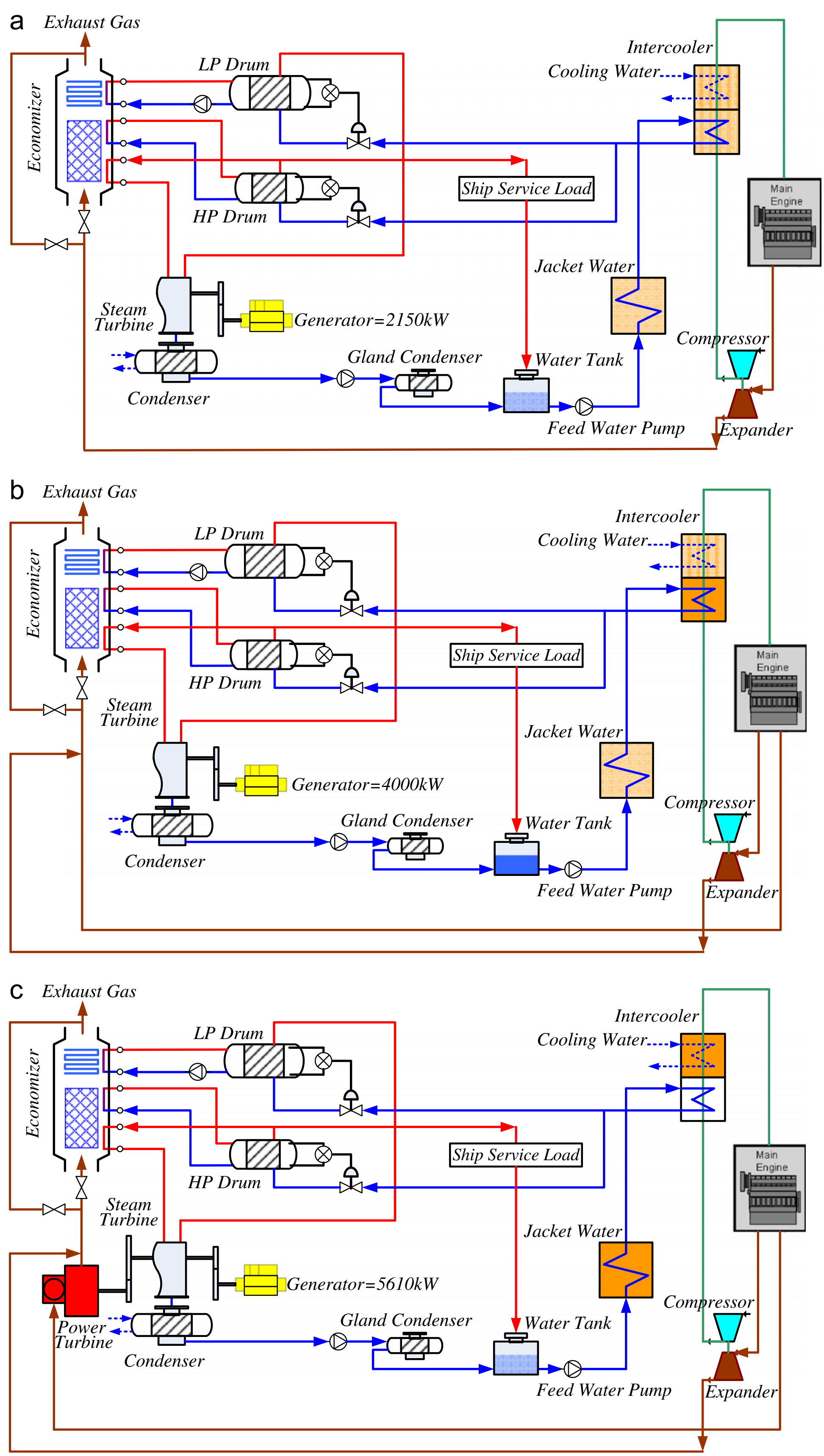

Fig. 12. (a) Dural pressure boiler, (b) System diagram with exhaust gas bypass and (c) System diagram with power turbine.

studied in connection to various local parameters such as capacity and type of energy available in low cost, WHR plant size and seawater temperature. Furthermore, the optimum design depends much on the type of vessel and its main operating conditions.

MAN, ABB, Mitsubishi heavy industries, Wartsila and Cummins have investigated various combined CHP (combined heat and power) systems on ships. Some CHP systems that have been already used on ships will be introduced in the following. These CHP systems are all improved and get their rapid development based on the basic turbo compound system (TCS) configuration. Comparison of generated output for different CHP systems was carried out by Mitsubishi heavy industries Ltd. This investigation is based on 12K98ME-C type engine of 69,900 kW.

Fig. 12(a) presents a basic compound WHR system with a turbocharger and a dual pressure multi-stage turbine. Exhaust gas from turbochargers outlet is 588,800 kg/h $\times$ 248.5 °C, which then flows into the exhaust boiler. The steam turbine consists of both high pressure (HP=7.5 kPa) and low pressure (LP=4.0 kPa). Heat is also recovered from the jacket cooling water and scavenging air cooler to preheat feed water before it enters into the boiler. Steam for the turbo generator is evaporated and superheated in two pressure stages. Each stage comprises a forced circulation evaporator, a steam separation drum, and a super-heater. The steam output is boosted by preheating the feed water in two stages by means of other main engine waste heat sources, namely jacket water and scavenging air. It is worth mentioning that the operating pressure of the LP section is selected and adjusted suitably high to avoid condensation of sulfuric acid that would immediately create excessive fouling problems and also corrosion damages on a longer term.

Due to the high efficiency of the turbocharger, the energy reclaimed from the turbine surpasses the requirement of compressor and will be wasted if all the exhaust gas goes through the turbocharger. Therefore, the design of an exhaust gas bypass protects the energy from waste in the condition of fulfilling the power of the turbocharger as shown in Fig. 12 (b). The exhaust gas bypass remains at the same temperature with that of the manifold outlet (assumed to have no energy loss), which will mix with that from the expander of the turbocharger. The temperature of the exhaust gas at the inlet of economizer is 309 °C, compared to 248.5 °C which is obtained without the bypass design. The energy recovered by the steam turbine generator is higher, 4000 kW compared to 2150 kW.

An additional power turbine is introduced in this configuration. As shown in Fig. 12 (c), the exhaust gas bypass is used to drive a power turbine and then mix this exhaust gas with that from the turbocharger outlet. In this system, the output power of power turbine and steam turbine is 2450 kW and 3160 kW respectively. Both of them drive a generator, of which the total electricity of which reaches 5610 kW. The overall efficiency will increase by 8.03.

These three different systems give an optimization process. It is evident that the last one is the most promising system and gives the highest efficiency compared to the others. However, we can find that the hot steam from the outlet of steam turbine is directly connected to a condenser, in which a huge amount of energy is wasted. There is still potential for alternative technologies such as fresh water generation to recover more of the waste heat. If the hot steam gets further utilized before the condenser, a higher overall efficiency will be reached.

MAN Diesel & Turbo Corporation [37] also investigates application of WHR systems on ships. It was shown that different recovery ratios were presented in the WHR system configurations. MAN proposed that turbo compound system with power turbine and generation (TCS-PTG) shuts down below 50 engine load. Both power turbine and turbocharger utilize the energy taken from the main engine exhaust gas. The maximum power output of WHR system can reach 10 and the overall efficiency can reach 55 when both power turbine and steam turbine are applied simultaneously [105].

The Emma Maersk's Wartsila-Sulzer RTA96-C turbocharged two-stroke diesel ship engine is the most powerful and the most efficient prime-mover of super ships in the world today. Comparisons of energy flux and fuel consumption between engine with and without the WHR system are shown in Fig. 13. For a combined system whose main engine is Wartsila-Sulzer RTA96-C, the use of the power turbine increases the power produced by the system by 2211 kW, i.e. by 4.77 for the 90 main engine load,

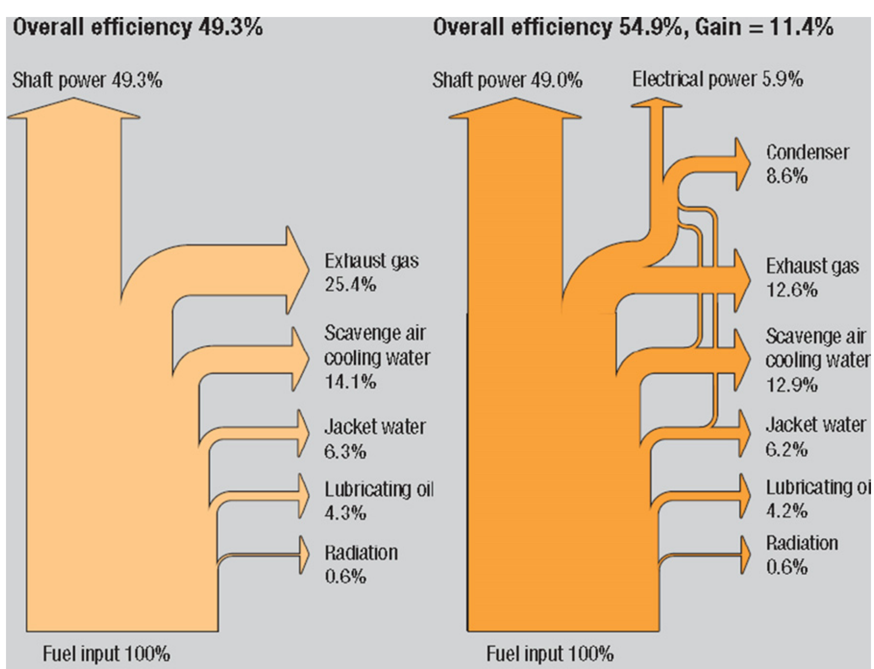

Fig. 13. Comparisons of energy flux and fuel consumption between engine with and without WHR system [115].

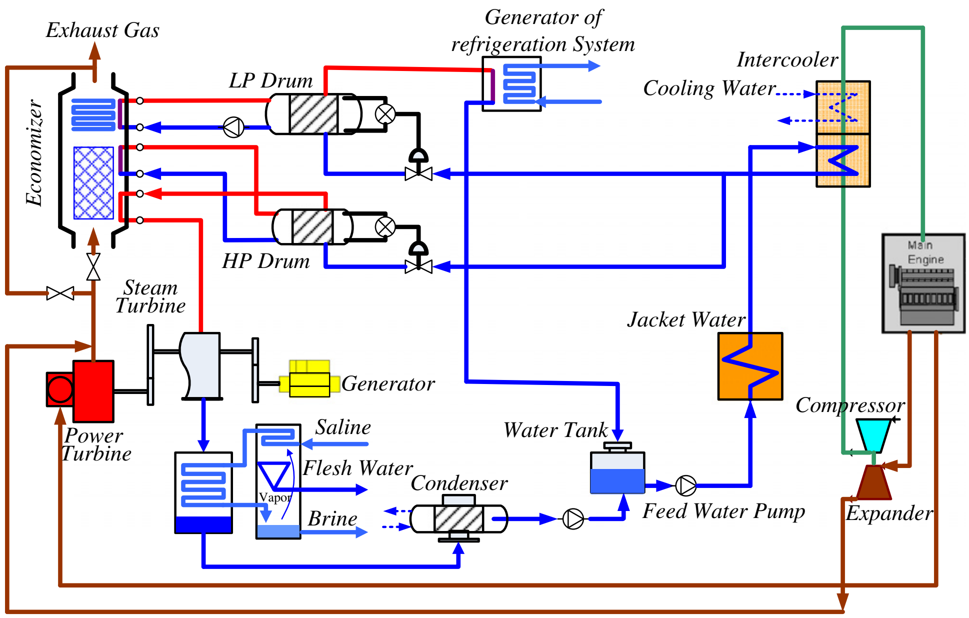

Fig. 14. A proposed novel system.

and decreases the specific fuel consumption by 4.6, as compared to the classical propulsion system. Introducing a steam turbine to the combined system increases its power by 11.4 and decreases its specific fuel consumption by 10.2. The efficiency of the combined propulsion system increases from 49.3 to 55.49 [2,115]. The annual operating cost for the main auxiliary engines would be US$19.54 million without a WHR plant and US$17.29 million with a high-efficiency WHR plant. There would be annual saving of US$2.25 million. As the complete high-efficiency WHR plant and its installation would call for an investment cost of about US$9.5 million, this would have an expected payback time of less than five years. Therefore, WHR plants can lead to considerable financial benefits over a ship's life especially for the increasing fuel prices nowadays.

4. Conclusion and future work

A detailed literature survey of WHR technologies based on waste heat aboard ships was performed. The aim is to provide comprehensive information about WHR for better improvement both in fuel consumption and emission. These technologies include turbine, refrigeration, Rankine cycle, desalination and combined cycle systems using more than two of these WHR technologies. Turbine technology have been approved and widely adopted on ships. From the aspect of theory, refrigeration is available to provide cooling by reclaiming waste heat. However, literature about its application on ship has not been reported yet. Rankine cycle and desalination technologies are already mature, which are convenient to obtain both the heat source and the cooling source. Although combined cycle systems are not widely used, these will receive more attention for their significant development potential to achieve higher thermal efficiency and to alleviate the atmosphere pollution problem. It will be a main research orientation of WHR technology.

Based on the technology and systems discussed above, a new system (Fig. 14) is developed for the purpose of further utilization of waste heat. For the configuration in Fig. 12(c), the steam flow from the outlet of the steam turbine is still higher than 100 °C, which will lead to a large amount of waste to be emitted into the atmosphere directly. However it can be reused because it can fulfill requirement of sorption refrigeration as heat source. The schematic of a new WHR combined system is shown. As the corrosion problem is usually caused by condensation during the expansion process in steam turbine, the low pressure steam from the exhaust economizer is utilized for adsorption refrigeration rather than that flowing into the steam turbine. The performance of this system will be investigated in the future work.

This paper will be useful for the researchers in WHR technologies to make effective decisions and generate more ideas. Thus the paper explicitly points out WHR technologies available on ships where there is potential for future research.

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References

[1] Eyringer V, Köhler HW, Lauer A, Lemper B. Emissions from international shipping: 2. Impact of future technologies on scenarios until 2050. Journal of Geophysical Research 2005;110:D17306.
[2] Dzida Marek, Mucharski Janusz. On the possible increasing of efficiency of ship power plant with the system combined of marine diesel engine, gas turbine and steam turbine in case of main engine cooperation with the gas turbine fed in parallel and the steam turbine. Polish Maritime Research 2009;1(59)(vol. 16):47–52.
[3] Jaichandar S, Annamalai K. Effects of open combustion chamber geometries on the performance of pongamia biodiesel in a DI diesel engine. Fuel 2012;98:272–9.
[4] Park S. Optimization of combustion chamber geometry and engine operating conditions for compression ignition engines fueled with dimethyl ether. Fuel 2012;97:61–71.
[5] Gan S, Ng HK, Pang KM. Homogeneous Charge Compression Ignition (HCCI) combustion: implementation and effects on pollutants in direct injection diesel engines. Applied Energy 2011;88:559–67.
[6] Yao M, Zheng Z, Liu H. Progress and recent trends in homogeneous charge compression ignition (HCCI) engines. Progress in Energy and Combustion Science 2009;35:398–437.
[7] Zheng M, Reader GT. Energy efficiency analyses of active flow aftertreatment systems for lean burn internal combustion engines. Energy Conversion and Management 2004;45:2473–93.

[8] Park C, Kim S, Kim H, Moriyoshi Y. Stratified lean combustion characteristics of a spray-guided combustion system in a gasoline direct injection engine. Energy 2012;41:401–7.
[9] Lu X, Shen Y, Zhang Y, Zhou X, Ji L, Yang Z. Controlled three-stage heat release of stratified charge compression ignition (SCCI) combustion with a two-stage primary reference fuel supply. Fuel 2011;90:2026–38.
[10] Knecht W. Diesel engine development in view of reduced emission standards. Energy 2008;33:264–71.
[11] Wartsila LTD. Solution for merchant vessels; 2010.
[12] Scappin F, Stefansson SH, Haglind F, Andreasen A, Larsen U. Validation of a zero-dimensional mode l for prediction of NO_x and engine performance for electronically controlled marine two-stroke diesel engines. Applied Thermal Engineering 2012;37:344–52.
[13] He M, Zhang X, Zeng K, Gao K. A combined thermodynamic cycle used for waste heat recovery of internal combustion engine. Energy 2011;36:6821–9.
[14] MAN B&W Diesel LTD. Thermo Efficiency System (TES) for reduction of fuel consumption and CO_2 emission. Copenhagen, Denmark: MAN B&W Diesel A/S, July 2005.
[15] Park S, Heo J, Yu BS, Phee SH. Computational analysis of ship's exhaust-gas flow and its application for antenna location. Applied Thermal Engineering 2011;31:
1689–702.
[16] Moldanova J, Fridell E, Popovicheva O, Demirdjian B, Tishkova V, Faccinetto A. Characterisation of particulate matter and gaseous emissions from a large ship diesel engine. Atmospheric Environment 2009;43:2632–41.
[17] Bidini G, Maria F, Generosi M. Micro-cogeneration system for a small passenger vessel operating in a nature reserve. Applied Thermal Engineering 2005;25:851–65.
[18] Tien W-K, Yeh R-H, Hong J-M. Theoretical analysis of cogeneration system for ships. Energy Conversion and Management 2007;48:1965–74.
[19] Rigby GR, Hallegraeff GM, Sutton C. Novel ballast water heating technique offers cost-effective treatment to reduce the risk of global transport of harmful marine organisms. Marine Ecology Progress Series 1999;191:289–93.
[20] Thombare DG, Verma SK. Technological development in the Striling cycle engines. Renewable and Sustainable Energy Reviews 2008;12:1–38.
[21] Cullen B, McGovern J. Energy system feasibility study of an Otto cycle/Striling cyclehybrid automotive engine. Energy 2010;35:1017–23.
[22] Eriksson L, Nielsen L, Brugard J, Bergstrom J, Pettersson F, Andesson P. Modeling of a turbocharger SI engine. Annual Reviews in Control 2002;26:129–37.
[23] Weerasinghe WMSR, Stobart RK, Hounsham SM. Thermal efficiency improvement in high output diesel engines a comparison of a Rankine cycle with turbo-compounding. Applied Thermal Engineering 2010;30:2253–6.
[24] Heywood JB. Internal combustion engine fundamentals. New York: McGraw-Hill; 1988 p. 248–70.
[25] Baines Nick, Wygant Karl D, Dris Antonis. The analysis of heat transfer in automotive turbochargers. ASME GT2009-59353.
[26] Karabektas M. The effects of turbocharger on the performance and exhaust emissions of a diesel engine fuelled with biodiesel. Renewable Energy 2009;34:989–93.
[27] Theotokatos GP. A modelling approach for the overall ship propulsion plant simulation. In: Proceedings of the 6th WSEAS international conference on system science and simulation in engineering. Venice, Italy; November 21–23, 2007.
[29] Stefanopoulou A, Smith R. Maneuverability and smoke emission constraints in marine diesel propulsion. Control Engineering Practice 2000;8:1023–31.
[30] MAN diesel & turbo LTD. VTA-variable turbine area. Augsburg, Germany: MAN Diesel & Turbo 86224. 〈www.mandieselturbo.com〉.
[31] Copeland CD, Newton P, Martinez-Botas R, Seiler M. The effect of unequal admission on the performance and loss generation in a double-entry turbocharger turbine. Journal of Turbo Machinery 2012;134:021004–15.
[32] Romagnoli A, Matinez-Botas RF, Rajoo S. Steady state performance evaluation of variable geometry twin-entry turbine. International Journal of Heat and Fluid Flow 2011;32:477–89.
[33] Xianfei Ding, Bin Xu. Study on regulating law of two-stage turbo charger system of piston aircraft engine. Procedia Engineering 2011;17:581–6.
[34] Galindo J, Serrano JR, Climent H, Varnier O. Impact of two-stage turbocharging architectures on pumping losses of automotive engines based on an analytical model. Energy Conversion and Management 2010;51: 1958–1969.
[35] Hopmann Ulrich, Marcelo C. Algrain diesel engine electric turbo compound technology. SAE paper 2003-01-2294.
[37] MAN diesel & turbo LTD. MAN diesel & turbo technology boosts efficiency. MAN diesel & turbo SE Teglholmsgade 41 DK-2450 Copenhagen SV DENMARK; June 2011. 〈www.mandieselturbo.com〉.
[38] Wang SG, Wang RZ. Recent developments of refrigeration technology in fishing vessels. Renewable Energy 2005;30:589–600.
[39] Manzela AA, Hanriot SM, Gomez LC, Sodre JR. Using engine exhaust gas as energy source for an absorption refrigeration system. Applied Energy 2010;87:1141–8.
[40] Kececiler A, Acar H, Dogan A. Thermodynamic analysis of the absorption refrigeration system with geothermal energy: an experimental study. Energy Conversion and Management 2000;41:37–48.
[41] Fernandez-Seara J, Vales A, Vazquez M. Heat recovery system to power an onboard NH_3–H_2O absorption refrigeration plant in trawler chiller fishing vessels. Applied Thermal Engineering 1998;18:1189–205.
[42] Maloney JD, Robertson RC. Thermodynamic study of ammonia-water power cycles. Oak Ridge (TN): Oak Ridge National Laboratory; 1953.
[43] Kalinowski P, Hwang Y, Radermacher R. Waste heat powered absorption refrigeration system in the LNG recovery process. In: Proceedings of the 9th international sorption heat pump conference. Seoul, Korea; 2008.
[44] Erickson DC. Extending the boundaries of ammonia absorption chillers. ASHRAE Journal 2007;49(4):32–5.
[45] Srikhirin P, Aphornratana S, Chungpaibulpatana S. A review of absorption refrigeration technologies. Renewable and Sustainable Energy Reviews 2001;5(4):343–72.
[46] Little Adrienne, Garimella Srinivas. Comparative assessment of alternative cycles for waste heat recovery and upgrade. In: Proceedings of the ASME 2009 3rd international conference of energy sustainability. ES2009-90023.
[47] Fan Y, Luo L, Souyri B. Review of solar sorption refrigeration technologies: development and applications. Renewable and Sustainable Energy Reviews 2007;11:1758–75.
[48] Hong Daliang, Chen Guangming, Tang Limin, He Yijian. A novel ejector–absorption combined refrigeration cycle. International Journal of Refrigeration 2011;34:159–60.
[49] Hong D, Tang L, He Y, Chen G. A novel absorption refrigeration cycle. Applied Thermal Engineering 2010;30:2045–50.
[50] Garimella S, Brown AM, Nagavarapu AK. Waste heat driven absorption/vapor-compression cascade refrigeration system for megawatt scale, high-flux, low-temperature cooling. International Journal of Refrigeration 2011. http://dx.doi.org/10.1016/j.ijrefrig.2011.05.017.
[51] Misra RD, Sahoo PK, Sahoo S, Gupta A. Thermoeconomic optimization of a single effect water/LiBr vapour absorption refrigeration system. International Journal of Refrigeration 2003;26:158–69.
[52] Kizilkan O, Sencan A, Kalogirou A. Thermoeconomic optimization of a LiBr absorption refrigeration system. Chemical Engineering and Processing 2007;46:1376–84.
[53] Rubio-Maya C, Pacheco-Ibarra JJ, Belman-Flores JM, Galvan-Gonzalez SR, Mendoza-Covarrubias C. NLP model of a LiBr–H_2O absorption refrigeration system for the minimization of the annual operating cost. Applied Thermal Engineering 2012;37:10–8.
[54] Misra RD, Sahoo PK, Gupta A. Thermoeconomic optimization of a LiBr/H_2O absorption chiller using structural method. Journal of Energy Resources Technology 2005;127(2):119–24.
[55] Srivastava NC, Eames IW. A review of adsorbents and adsorbates in solid–vapour adsorption heat pump systems. Applied Thermal Engineering 1998;18:707–14.
[56] Ruthven DM. Principles of adsorption and adsorption processes. New York: Wiley; 1984.
[57] Suzuki M. Adsorption for energy transport. Japan: Adsorption Engineering; 1980.
[58] Liu Y, Leong KC. The effect of operating conditions on the performance of zeolite/water adsorption cooling systems. Applied Thermal Engineering 2005;25:1403–18.
[59] Solmus I, Kaftanoglu B, Yamali C, Baker D. Experimental investigation of a natural zeolite–water adsorption cooling unit. Applied Energy 2011;88:4206–13.
[60] Zhang LZ. Design and testing of an automobile waste heat adsorption cooling system. Applied Thermal Engineering 2000;20:103–14.
[61] Wang RZ, Wu JY, Xu YX, Wang W. Performance researches and improvements on heat regenerative adsorption refrigerator and heat pump. Energy Conversion and Management 2001;42:233–49.
[62] Wang LW, Wang RZ, Lu ZS, Chen CJ, Wang K, Wu JY. The performance of two adsorption ice making test units using activated carbon and a carbon composite as adsorbents. Carbon 2006;44:2671–80.
[63] Wang DC, Xia ZZ, Wu JY. Design and performance prediction of a novel zeolite–water adsorption air conditioner. Energy Conversion and Management 2006;47:590–610.
[64] Lu ZS, Wang RZ, Wu JY, Chen CJ. Performance analysis of an adsorption refrigerator using activated carbon in a compound adsorbent. Carbon 2006;44:747–52.
[65] Wang RZ. Performance improvement of adsorption cooling by heat and mass recovery operation. International Journal of Refrigeration 2001;24: 602–11.
[66] Shelton SV, Wepfer WJ. Solid–vapor heat pump technology. In: Proceedings of the IEA heat pump conference. Tokyo; 1990. p. 525–35.
[67] Critoph RE. Forced convection adsorption cycle with packed bed heat regeneration. International Journal of Refrigeration 1999;22:38–46.
[68] Ziegler F, Brandl F, Volkel J, Alefeld G. A cascading two-stage sorption chiller system consisting of water–zeolite high temperature stage and a water–LiBr low temperature stage. Absorption Heat Pump Congress Paris 1985: 231–8.
[69] Liu Y, Leong KC. Numerical modeling of a zeolite/water adsorption cooling system with non-constant condensing pressure. International Communications in Heat and Mass Transfer 2008;35:618–22.
[70] Tatlıer M, et al. A novel approach to enhance heat and mass transfer in adsorption heat pumps using the zeolite–water pair. Microporous and Mesoporous Materials 1999;27:1–10.

[71] Wang RZ, Wu JY, Xu YX, Wang W. Performance researches and improvements on heat regenerative adsorption refrigerator and heat pump. Energy Conversion and Management 2001;42:233–49.
[72] Tamainot-Telto Z, Critoph RE. Adsorption refrigerator using monolithic carbon–ammonia pair. International Journal of Refrigeration 1997;20(2): 146–55.
[73] Lu YZ, Wang RZ, Jianzhou S, Xu YX, Wu JY. Practical experiments on an adsorption air conditioner powered by exhausted heat from a diesel locomotive. Applied Thermal Engineering 2004;24:1051–9.
[74] Wang LW, Bao HS, Wang RZ. A comparison of the performances of adsorption and resorption refrigeration systems powered by the low grade heat. Renewable Energy 2009;34:2373–9.
[75] Li TX, Wang RZ, Oliveira RG, Kiplagat JK, Wang LW. A combined double-way chemisorptions refrigeration cycle based on adsorption and resorption processes. International journal of Refrigeration 2009;32:47–57.
[76] Hendricks TJ, Lustbader JA. Advanced thermoelectric power system investigations for light-duty and heavy-duty vehicle applications: part I. In: Proceedings of the 21st international conference on thermoelectrics. IEEE Catalogue; 2002. p. 381–6.
[77] Rowe DM, Min G, Williams SGK, Aoune A, Matsuura K, Kuznetsov VL, et al. Thermoelectric recovery of waste heat—case studies. In: Proceedings of the 32nd international energy conversion engineering conference, vol. 2; 1997. p. 1075–9.
[78] Hendricks Terry, Choate William T. Engineering scoping study of thermoelectric generator systems for industrial waste heat recovery; November 2006.
[79] Kajikawa T.. Present state of R&D on thermoelectric technology in Japan. In: Proceedings of the 20th international conference on thermoelectrics. IEEE catalogue; 2001. p. 49–56.
[80] Bass JC, Elsner NB Leavitt FA. Performance of the 1 kW thermoelectric generator for diesel engines. In: Proceedings of the international conference on thermoelectric. Kansas City, Kansas, USA; 1994.
[81] Michael Rowe David. Thermoelectric waste heat recovery as a renewable energy source. International Journal of Innovations in Energy Systems and Power 2006;1(1).
[82] Hendricks Terry, Choate William T. Engineering scoping study of thermoelectric generator systems for industrial waste heat recovery; November 2006.
[83] Gequn Shu, Zhao J, Tian H, Wei H, Liang X, Yu G. Theoretical analysis of engine waste heat recovery by the combined thermo-generator and organic Rankine cycle system. SAE paper 2012-01-0636.
[84] Haidar Jihad G, Ghojel Jamil I. Waste heat recovery from the exhaust of low-power diesel engine using thermoelectric generators. In: Proceedings of the 20th international conference on thermoelectrics; 2001.
[85] Michael Rowe David. Thermoelectric waste heat recovery as a renewable energy source. International Journal of Innovations in Energy Systems and Power 2006;1(1).
[86] Wang T, Zhang Y, Peng Z, Shu G. A review of researches on thermal exhaust heat recovery with Rankine cycle. Renewable and Sustainable Energy Reviews 2011;15:2862–71.
[87] Ringler J, Seifert M, Guyotot V, Hubner W. Rankine cycle for waste heat recovery of IC engines. SAE paper 2009-01-0174; 2009.
[88] Hung TC, Shai TY, Wang SK. A review of organic Rankine cycles (ORCs) for the recovery of low-grade waste heat. Energy 1997;22(7):661–7.
[89] Bombarda P, Invernizzi CM, Pietra C. Heat recovery from diesel engines: a thermodynamic comparison between Kalina and ORC cycles. Applied Thermal Engineering 2010;30:212–9.
[90] Larjola J. Electricity from industrial waste heat using high-speed organic Rankine cycle (ORC). International Journal of Production Economics 1995;41:227–35.
[91] Gewald D, Konstantinos S, Sotirios K, Hartmut S. Waste heat recovery from a landfill gas-fired power plant. Renewable and Sustainable Energy Reviews 2012;16:1779–89.
[93] Leibowitz H, Smith IK, Stosic N. Cost effective small scale ORC systems for power recovery from low grade heat sources. In: IMECE 2006-14284, ASME international mechanical engineering congress and exposition. Chicago, Illinois, USA; November 5–10, 2006.
[94] Brasz Joost J, Biederman BP, Holdmann G Power Production from a moderate-temperature geothermal resource. GRC annual meeting. Reno, NV, USA; September 25–28th; 2005.
[95] Liu B-T, Chien K-H, Wang C-C. Effect of working fluids on organic Rankine cycle for waste heat recovery. Energy 2004;29:1207–17.
[96] Hung TC. Waste heat recovery of organic Rankine cycle using dry fluids. Energy Conversion and Management 2001;42:539–53.
[97] Yamamoto T, Furuhata T, Arai N, Mori K. Design and testing of the organic Rankine cycle. Energy 2001;26:239–51.
[98] Khawaji AD, Kutubkhanah IK, Wie J-M. Advances in seawater desalination technologies. Desalination 2008;221:47–69.
[99] Borsani R, Rebagliati R. Fundamentals and costing of MSF desalination plants and comparison with other technologies. Desalination 2005;182:9–37.
[100] Elshorbagy Walid, Abdulkarim Mohamed. Chlorination byproducts in drinking water produced from thermal desalination in United Arab Emirates. Environmental Monitoring and Assessment 2006;123:313–31.
[101] Buros OK. The U.S.A.I.D. desalination manual. Gainesville, Florida: International Desalination and Environmental Association; 1980.
[102] Ghirardo Federico, Santin Marco, Trave rso Alberto, Massardo Aristide. Heat recovery options for onboard fuel cell systems. International Journal of Hydrogen Energy 2011;36:8134–42.
[103] Sommariva C, Venkatesh R. MSF desalination. MEDRC Newsletter 2002;18:1–3.
[104] IDA Desalination Yearbook. Water desalination report. Topsfield (MA), USA: Global Water Intelligence and International Desalination Association; 2006–2007.
[105] Al-Juwayhel F, Dessouky EI, Ettouney H. Analysis of single-effect evaporator desalination systems combined with vapor compression heat pumps. Desalination 1997;114:253–75.
[106] Borsani R, Rebagliati R. Fundamentals and costing of MSF desalination plants and comparison with other technologies. Desalination 2005;182:9–37.
[107] Karagiannis IC, Soldatos PG. Water desalination cost literature: review and assessment. Desalination 2008;223:448–56.
[108] Eltawil MA, Zhengming Z, Yuan L. A review of renewable energy technologies integrated with desalination systems. Renewable and Sustainable Energy Reviews 2009;13:2245–62.
[109] Al-Sahali M, Ettouney H. Developments in thermal desalination processes: design, energy, and costing aspects. Desalination 2007;214:227–40.
[110] Ophir A, Lokiec F. Advanced MED process for most economical sea water. Desalination 2005;182:81–192.
[111] Ettouney Hisham. Design and analysis of humidification dehumidification desalination process. Desalination 2005;183:341–52.
[112] Morin OJ. Design and operating comparison of MSF and MED systems. Desalination 1993;93(1–3):69–109.
[113] Vlachos GTh, Kaldellis JK. Application of gas-turbine exhaust gases for brackish water desalination: a techno-economic evaluation. Applied Thermal Engineering 2004;24:2487–500.
[114] Ebensperger U, Isley P. Review of the current state of desalination: water policy working paper 2005–008. Georgia Water Planning and Policy Center; 2005. 〈http://www.h2opolicycenter.org/wp2005.shtml〉 [accessed 15.09.07].
[115] Wartsila Corporation. Waste heat recovery. Leaflet; 2007. p. 6.
[116] Evans LR, Miller JE. Sweeping gas membrane desalination using commercial hydrophobic hollow fiber membranes. SAND report. SAND 2002-0138. Unlimited release. Printed January; 2002.
[117] Karytsas. Mediterranean conference on renewable energy sources for water production, European commission. EURORED Network. Santorini, Greece: CRES; 1996. p. 128–31.